CONTINUOUS MONITORING SYSTEMS FOR HIGH SPEED GEARING

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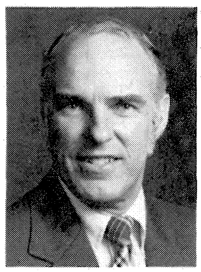

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Mr. Mitchell's extensive experience includes maintenance engineering and engineering supervision for a major international oil company followed by over ten years consulting engineering for a wide variety of clients in the oil, petrochemical, power generating and maritime industries. For three years prior to forming his own company, he was Director of Marketing and Technical Services for the industrial machinery monitoring division of a major instrument manufacturer.

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ABSTRACT

The most effective methods for monitoring the health of high speed gearing is currently generating more interest and discussion than any other aspect of machinery health monitoring.

Both structurally and dynamically, a gear is quite different from the machines coupled to it. As a result, monitoring gear health often requires methods which differ from those that have proven exceptionally effective on other types of critical machinery.

This paper describes the dynamics of high speed gears that apply to and influence health monitoring. Various types of monitoring systems that have been proposed and utilized are discussed, followed by the advantages and limitations of each system. Actual case histories are cited to illustrate many of the elements of gear health monitoring.

The paper concludes with recommended health monitoring systems for several classes of gears. Each system is described in detail along with the reasons for selecting the specific method.

INTRODUCTION

The best method or methods for monitoring the condition of gears is currently a controversial subject, virtually guaranteed to spark lively discussion. With the use of more and larger geared machinery strings, it is appropriate to examine this subject in the same depth that has been devoted to monitoring steam turbines and centrifugal compressors.

Dynamically, a gear is far different from either a steam turbine or centrifugal compressor. As a direct result, the optimum methods for measuring gear condition may well vary from the methods that have proven so effective on turbines and compressors.

Before proceeding further, we should define condition monitoring as it will be used in this paper. For our purposes, condition monitoring is defined as a method or methods to judge the overall condition of operating equipment.

Condition monitoring has two aspects, measuring variables that are most indicative of overall condition followed by displaying those measurements in some easily understood form.

Condition monitoring is focused on protecting the machine, the process and people who may be nearby.

An effective condition monitoring system must provide an accurate and reliable measure of current condition which, in extremis, can be used for automatic shutdown.

Condition monitoring should be able to call attention to changes in condition at as early a stage as possible. The latter, predictive, aspect of condition monitoring differs from diagnostics. Diagnostics are directed toward identifying the specific cause of a change in condition as well as the specific components affected. Although the distinction may be a fine one, it is the author's opinion that diagnostics, or problem definition, are not required for effective condition monitoring even though the basic information used for diagnostics may be derived from condition monitoring.

As you, the reader, will observe during the course of this paper, the emphasis is placed on measurements and methods of conditioning raw data. Our objective is to derive and present information which accurately depicts gear condition and is highly responsive to changes in condition.

To meet our objective we'll develop methods for most efficiently monitoring gears in three classifications. First, however, we will describe the dynamic characteristics of gears, outline some common gear problems and their symptoms and finally discuss the advantages and limitations of some of the methods commonly employed to judge condition.

As one final note, our discussion will be restricted to parallel shaft increasers and reducers; right angle and epicyclic gears are not included even though the principles can certainly be applied.

THE DYNAMIC CHARACTERISTICS OF A GEAR

Dynamically, a speed increasing or reducing gear is significantly different than a centrifugal compressor or steam turbine. The bearing load on two bearing compressors and turbines is generally at or close to one half the rotor weight. In contrast, the torque force developed across meshing gears, produces bearing forces which are several times the rotor weight. As a result, parallel shaft gear bearings are highly loaded and operate at higher temperatures than radial bearings on turbines and compressors. Due to the large bearing loads, subsynchronous instability is seldom a problem on parallel shaft gears.

Figure 1 shows calculated full load bearing parameters developed for an up mesh pinion. Note that the vertical, Y, bearing load is 8.5 times the rotor weight — and in the opposite direction, toward the top of the bearing! In recognition of this load distribution, you will note that the pressure dam is located in the bottom half shell of the bearing.

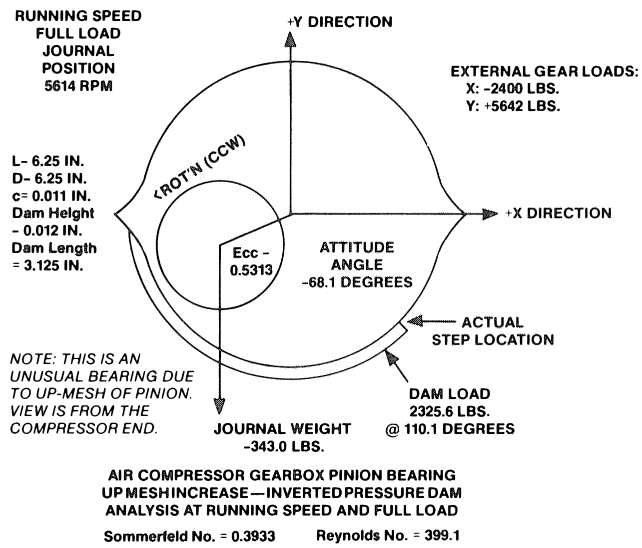

Figure 1. Full Load Bearing Parameters for an Up-Mesh Pinion.

Digressing for a moment, it should be pointed out that the final, full load, position of the pinion in an up mesh parallel shaft increaser may be above the horizontal centerline of the bearing. Although it didn't occur in this specific case which was confirmed by measurement, a large shift in position from no load to full load can lead to severe problems.

Whenever a journal must move through more than 70° - 80° from its position at rest in the bottom of a bearing, to its final full load position, the bearing shell must be clocked so it will provide full support throughout the shift in shaft position. Failure to recognize this situation has caused catastrophic bearing failures at an intermediate load when shaft position coincided with the oil inlet groove located at the bearing's horizontal split. The small bearing area present along the horizontal split was unable to support the large load applied and the bearing quickly failed.

Figure 2 illustrates the undamped critical speed map for the same gear. Note the increase in both X and Y bearing stiffness with speed. Here again torque produces a far greater change in stiffness than will be found on either a turbine or centrifugal compressor.

The large forces just cited exert a powerful restraint on a gear shaft. Similarly these large forces restrict relative motion between the shaft and its bearing. This may be one reason behind reports of gear damage and failures which were not

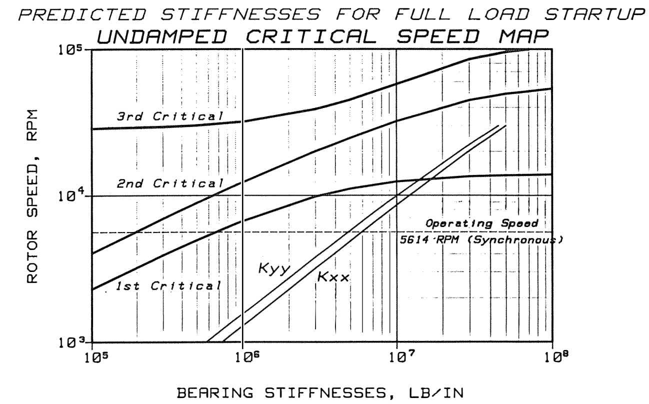

Figure 2. The Undamped Critical Speed Map for an Up-Mesh Pinion.

readily apparent on monitoring systems employing relative motion displacement transducers.

Although a great deal of effort goes into designing a smooth transfer of power from one gear to the other, there is always some variation in load as individual teeth come into and out of mesh. This variation in load produces the mesh frequency (number of teeth times shaft rotating frequency). The mesh frequency is generally the dominant sound generated by a gear and has always been used by experienced mechanics as a prime indicator of gear condition.

Much has been written regarding the use of the mesh frequency to evaluate tooth condition. Unfortunately it is difficult to generalize, for the amplitude at the mesh frequency is also affected by factors such as the number of teeth on a gear, and gear load.

As we will discuss more fully in the next section, there are some general observations that can be made regarding mesh and other high frequencies which may be very useful in condition monitoring. However, at the present time the use of high frequency vibration to gain an accurate indication of tooth wear and condition is largely a matter of experience and intuition.

TYPICAL GEAR PROBLEMS AND THEIR SYMPTOMS

Like all rotating machinery, gears are subject to imbalance and external misalignment. The symptoms of both problems are well known, discussed fully in the literature and generally easy to recognize.

As a point worth noting, it is often difficult to distinguish between imbalance and pitch line runout illustrated in Figure 3. Both produce excitation at the running frequency of the offending gear. Due to the high restraining force that is produced perpendicular to the plane passing through both shafts, vibration caused by imbalance or pitch line runout is generally greater in the plane containing the shafts.

The mesh frequency and its harmonics have been reported to contain information that defines tooth condition. [1] Unfortunately, normal variations in the amplitude at mesh frequency, illustrated in Figure 4, may obscure changing condition.

The signatures contained in Figure 4 were recorded from an accelerometer mounted on the gearbox of a marine steam turbine generator as load was reduced from full to no load. You will note that the amplitude at mesh frequency increased steadily from 2 g's at 1,000 KW to 10 g's at 200 KW. Since the

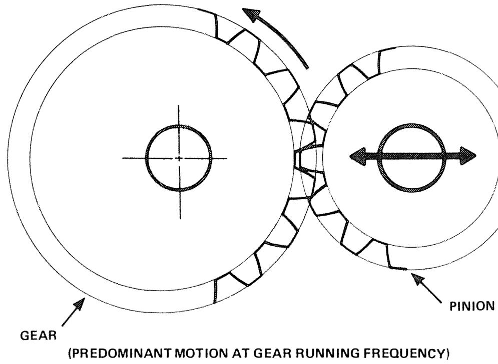

Figure 3. Pitch Line Runout.

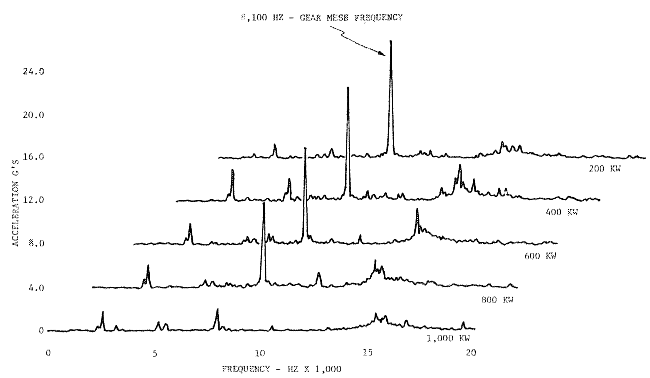

Figure 4. Turbine Generator Gear Acceleration Signatures Recorded During Load Reduction.

load change was accomplished over approximately 10 minutes on a turbine generator that had been operating continuously for several months, it's a safe assumption that mechanical conditions didn't change. Reference 2 contains a second example in which the amplitude at mesh frequency changed by a factor of 4 from no load to full load.

If the amplitude at mesh frequency and its harmonics can change due to a variety of factors that are unrelated to condition, is this a useful measurement to make? From the standpoint of early warning the answer is yes, however, one must be attuned to look for abnormal change.

The so called, "intermediate frequencies" have been mentioned as indicators of mechanical condition. [2] Since reference 2 was published the author has learned of numerous cases where an increase in amplitude at the intermediate frequencies was the earliest clear warning of a problem.

One particularly interesting problem, manifested by an increase in amplitude at the intermediate frequencies, produced rapid, heavy wear that caused measurable variation in tooth spacing. When the variation in tooth spacing was plotted, it was discovered that the wear pattern repeated the same number of times around the circumference of the gear as the intermediate frequency observed in the vibration spectrum! In this case some unidentified mechanism had caused the gear to "lock on" to a repetitive variation in tooth loading.

Whether the intermediate frequencies are produced by resonance of the gear elements excited by faulty mesh as proposed in reference 2, or related to some other phenomena, [3] their amplitude appears clearly related to conditions at mesh. As a result, including the intermediate frequencies into any scheme of condition, monitoring is clearly justified.

Thus far we have discussed overall problems with gears. Often however, a problem will be local in nature such as a bad tooth or one or more bad areas on one or both gears. A classic example occurred on a large marine reduction gear when a stone, being used to dress the gear teeth, was left on the gear as it was barred over. The stone, or more accurately pieces, passed through the mesh, badly damaging two localized areas on each gear. When a local defect is present, a strong amplitude variation at the hunting tooth frequency may be observed. A hunting tooth frequency (rotating speed of the gear divided by number of teeth on the pinion), is always too low to be heard directly. Rather it is generally observed as a clearly audible variation in the amplitude at mesh frequency.

Finally, we should mention manufacturing defects. We've already discussed the effect of pitch line runout. Apex runout on double helical gears will cause the gears to shuttle axially. Tooth spacing errors will cause a modulation of the gear mesh frequency nearly identical to torsional vibration.

PHILOSOPHY FOR GEAR CONDITION MONITORING

Based on experience, one can safely say that every gear problem will sooner or later affect vibration at low frequencies, around shaft rotating speeds. While mesh defects may show up much earlier at high frequencies, the high frequencies are affected by other factors such as changes in load. Thus, there are strong arguments for restricting machinery protection, particularly automatic shutdown, to low frequency information. This is not to eliminate the high frequencies from consideration in condition monitoring, but rather to think of them as predictive rather than protective.

Vibration information obtained from a non-contact displacement transducer is automatically limited to low frequencies below about 1,000 to 1,500 HZ. This limitation is not due in any way to a limitation of the transducer or its electronics, but rather due to the unbearable force needed to generate a measurable displacement at high frequencies.

An accelerometer, monitoring casing vibration, will produce a signal that is weighted in favor of the high frequencies. This is due to the frequency squared proportionality inherent in acceleration. As a result, acceleration signals must be conditioned prior to use for condition monitoring. Integrating an acceleration signal to velocity is a generally accepted method of enhancing the lower frequencies for condition monitoring.

High frequency components, such as gear mesh, are likely to make up a significant portion of the overall signal obtained from an accelerometer mounted on a gear, even after integration to velocity. If not handled properly, the simultaneous presence of both low and high frequency information may reduce the monitor's response to changes in condition. In addition, the presence of high frequencies in the signal being monitored may in time produce a loss of credibility among the people who must use and depend on the monitor to provide an accurate picture of gear condition.

A loss in credibility is particularly serious because the monitor may not be believed when it is needed most. One such situation is when the monitor first registers a change. When a change occurs, most will immediately attempt to confirm the change with a measurement taken at the same location with a portable instrument. If there is any significant variation between the value read on the monitor and the measurement obtained from the portable instrument, especially if the monitor reading is higher, the monitor is always suspect.

In fact, both measurements may be correct. If the monitor has a broader bandwidth than the portable instrument, and the gear is generating a significant amount of excitation at frequencies which are above the range of the portable instrument, the monitor should, and will, read higher. Unfortunately, many who have been confronted with a discrepancy between two presumably equivalent measurements have not recognized the cause for the discrepancy or its significance, with catastrophic results.

Whenever an instrument with a higher upper frequency response reads a larger value compared to an instrument with a lower response, the difference must be due to excitation which is within the passband of one instrument and outside the passband of the other. This, of course, assumes both instruments are properly calibrated and measuring the same variable at the same location. Unfortunately, few people who use machinery monitoring and analysis instruments have much of an appreciation for the passbands of various measurements and instruments.

On a gear, being able to distinguish between changes at high and low frequencies will convey a great deal of information regarding condition. If an abnormal increase in vibration occurs at high frequencies, it is likely an early symptom of a problem involving fatigue or a mesh anomaly. If such a change goes unrecognized, or is discounted because the measurement cannot be duplicated with a portable instrument, the early warning is lost and outright failure may result.

Fortunately, there is a method readily available to ensure changes among either low or high frequencies will be clearly recognized.

Earlier, we mentioned that the low frequencies could be thought of as protective while the high frequencies could be considered predictive. For the purposes of monitoring, the author recommends separating the lower, protective, frequencies by filtering the casing velocity signal to a passband from 10HZ to 1,000 HZ. This passband may have to be adjusted if it does not enclose the running speeds of both gear shafts. For example, the low frequency cutoff will have to be decreased to 1 HZ or 2 HZ for monitoring low speed equipment such as cooling tower and mixer gears.

The recommended passband will enclose all the frequencies required for protection and will be equivalent to the bandwidth of most portable instruments.

With a means to provide reliable, continuous machinery protection and ensure a casing vibration monitor and portable instrument will read about the same value, we next must address the question of displaying the high frequency, predictive, portion of the vibration signal for early warning.

Multiple path monitoring, first proposed in reference 4, divides the signal from a single accelerometer into two or more parallel paths for processing. One path will consist of the lower, protective frequencies separated as described in the previous paragraphs. Other paths are similarly selected with filters to enclose various frequencies of interest. The author recommends monitoring frequencies above 1,000 HZ in acceleration to take advantage of the frequency squared enhancement of the higher frequencies.

Figure 5 illustrates how a complex gear spectrum can be divided into two and three parallel paths for monitoring. The upper configuration divides the signal into two paths with the division between paths at 1,000 HZ. In this example, the lower path would be monitored in velocity, the upper path in acceleration.

The lower example in Figure 5 divides the signal into three paths for more discrimination. In this second example, the first split is made at 1,000 HZ. The second path extends from 1,000 HZ to just above the gear mesh frequency and is monitored in acceleration. The upper path includes the gear

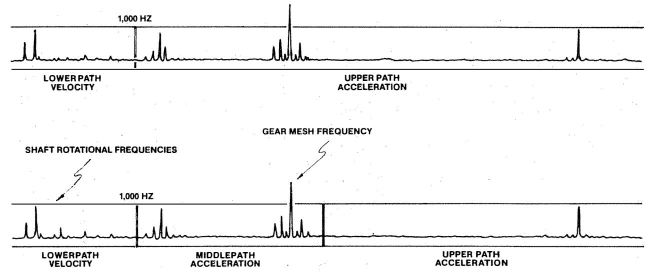

Figure 5. Dividing a Gear Vibration Spectrum for Multi Path Monitoring.

mesh frequency and two harmonics and is likewise monitored in acceleration.

You can see that this approach to gear monitoring is exceptionally flexible and may take many variations. One variation utilized a switchable filter to view a broad high frequency path or the gear mesh component along. [5] This could be taken a step further to monitor the ratio between the amplitude at gear mesh frequency and the remainder of the high frequency vibration.

It should be noted that the ratio just described, as well as the presence and strength of sidebands around the gear mesh frequency, have been proposed as measures of gear condition. [1, 6, 7] This was probably conceived recognizing that absolute amplitudes of the gear mesh frequency, its multiples and sidebands vary due to factors not related to gear condition.

To summarize, multiple path monitoring is highly advantageous when applied to gears. The separation of frequencies ensures a maximum response to a variety of changes in condition. Focusing each data path on specific characteristics ensures accurate, reliable machinery protection without compromising the monitor's ability to respond to changes in the high, predictive, frequencies. Multiple path monitoring is easy to understand and it minimizes the possibility of misinterpreting the significance of a change.

STRENGTHS AND LIMITATIONS OF TYPICAL GEAR CONDITION MONITORING SYSTEMS

Condition monitoring systems employing non-contact displacement pickups are well understood, automatically select the low frequency portion of the spectrum for the purposes of protection and will probably be employed for primary protection on both driver and driven machinery.

On the other hand, condition monitoring based on measurements made with non-contact shaft displacement sensors has several disadvantages on gears.

First, the large bearing forces found on gears tends to suppress relative motion. As discussed earlier, this factor reduces the ability of measurements made with non-contact displacement sensors to accurately represent condition and respond to changes in condition.

Next, displacement measurements will not respond at all to changes in condition whose symptoms appear at high frequencies. Thus, in certain situations, non-contact displacement measurements may provide little advance warning of an impending failure.

Possibly one or both of the foregoing factors have combined in several gear failures where a non-contact displacement monitoring system, in proper working order, reportedly did not provide any indication of a problem, even though symptoms of distress were clearly noted with other methods.

Finally, the non-contact condition monitoring system typically installed on a gear is relatively complex employing two X-Y pickup pairs at the coupling end of the input and output shafts.

A gear condition monitoring system employing one or more seismic accelerometers has the advantage of being able to view a broader range of characteristics. Additionally it is generally less complex and slightly less expensive than a comparable monitoring system based on non-contact displacement pickups.

More complex signal conditioning, including the necessity to limit the frequencies picked up by a typical acceleration sensor used for gear monitoring, might be considered disadvantages.

RECOMMENDED GEAR MONITORING SYSTEMS

For the purposes of this section, gears will be divided into three classes as follows:

- Critical, high speed special purpose gears
- Vital, special and general purpose gears
- General purpose gears requiring protection due to factors such as a remote and/or inaccessible location, unattended operations, etc.

Critical, High Speed Special Purpose Gears

Although the author considers seismic acceleration sensors superior for gear condition monitoring, some users are more comfortable with non-contact displacement probes and will require these sensors on critical gears.

When non-contact displacement probes are used for continuous condition monitoring on gears, an X-Y pair should be located at the coupling end of each shaft reading out on dual channel X-Y monitors per API Standard 670.

The author recommends two acceleration sensors mounted horizontally on a stiff section at the input and output shaft coupling end bearings. Dual path monitoring should be accomplished as a minimum on the signals from each acceleration sensor. For the reasons discussed earlier, the lower path should be monitored in velocity from 10 HZ to 1,000 HZ. The higher path should be monitored in acceleration from 1,000 HZ to the upper limit of the acceleration sensor, at least 10 KHZ, and higher if possible. In the event automatic shutdown is contemplated, dual voting logic requiring velocity levels in the lower, protective, paths of both monitors above their danger setpoint as a prerequisite for trip is highly recommended.

Bearing temperatures should be continuously monitored with temperature sensors embedded in the load zone of each radial bearing. Thrust temperature monitoring with embedded sensors is recommended, particularly on single helical gears where the thrust bearing may be heavily loaded.

Axial position monitoring should be considered for double helical gears where growth into the gear, combined with coupling lock-up, can overload one helix. Unmonitored non-contact axial position probes are recommended for assessing apex runout on new double helical gears and are often very handy for diagnostic purposes.

Vital, Special and General Purpose Gears

Gears in this category are adequately monitored with two acceleration sensors installed horizontally at the input and output shaft coupling end bearings. Monitoring may be single or dual path depending on the specific gear. Dual path monitoring should be employed on especially vital gears. Bearing temperature and axial position monitoring are not generally required.

General Purpose Gears Requiring Protection

In general, a single accelerometer mounted horizontally at the high speed coupled end bearing will provide all the information needed for protection. The signal should be integrated to velocity, filtered as described, or left broadband, in a continuous monitor equipped with two level warning alarms.

RECOMMENDATIONS FOR EFFECTIVE GEAR MONITORING

Accelerometer Location and Mounting

The ability of an accelerometer to sense machinery condition and respond to changes in condition is heavily dependent on its location and method of attachment. An accelerometer must be attached to a stiff section of the gear case where the vibration is related to mechanical condition rather than structural response.

If time is available, one can test several locations and orientations on an operating gear with an accelerometer and glue-on mounts. By comparing spectrum plots from each trial, the optimum accelerometer location is generally evident. If this type of test cannot be performed, experience indicates that mounting an accelerometer in the horizontal plane, at the coupling end bearing of the high speed shaft, will usually provide acceptable results. If a second accelerometer is to be used, it should be placed in a similar location at the coupling end bearing of the low speed shaft.

Mounting a permanently installed condition monitoring accelerometer below the horizontal split is advantageous. In this position, it doesn't have to be removed or disturbed when the machine is disassembled.

To prevent damage, accelerometers must be protected by either a cover or a rigid shield. In either case, care must be taken so the protective cover or shield does not excite the accelerometer.

It is generally quite difficult to obtain a sufficiently flat surface for mounting an accelerometer on installed equipment. In this situation, the mounting shown in Figure 6 is highly recommended. To construct the mounting illustrated in Figure 6, a bolt with a sufficiently large head is faced then drilled and tapped for the accelerometer mounting screw. A hole to receive the bolt is drilled and tapped at the location selected on the machine casing. The bolt is then screwed tightly into the hole in the machine casing and the accelerometer attached. Do

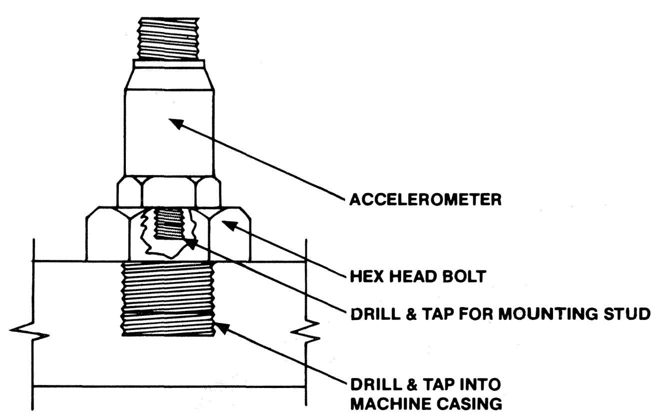

Figure 6. Recommended Accelerometer Mount.

not allow any of the body of the bolt to be exposed. The author has observed accelerometer mounts fabricated in this manner in a bending resonance due to excessive cantilevered length.

When an explosion proof enclosure is required, the mounting illustrated in Figure 7 should be considered. To ensure the best transmission path from machine casing to the accelerometer the bolt in Figure 6 is replaced by a machined adapter. One end of the adapter is drilled for the accelerometer mounting stud and threaded to mate with the female threads in a standard explosion proof enclosure. The opposite end is threaded to screw into the gear case as before.

It should be noted that the housing arrangement illustrated in Figure 1 of API Standard 678 is for mechanical protection only, it is not permissible as an explosion proof enclosure.

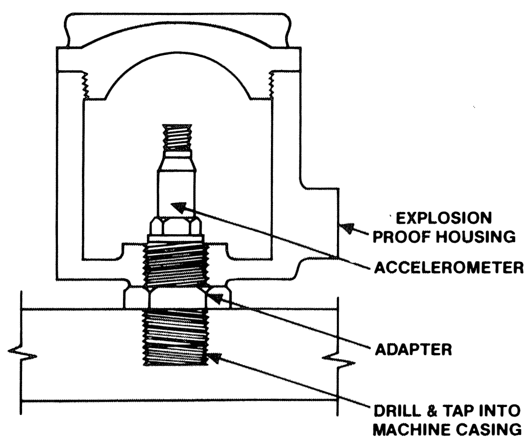

Figure 7. Recommended Method of Mounting an Accelerometer Inside an Explosion Proof Housing.

Considerations for Continuous Monitors

Continuous monitors with the features required by API Standards 670 and 678 will provide reliable protection for critical gears. Simpler monitors without some of the API features such as first out alarm indication and individual displays of the monitored values will provide acceptable protection for general purpose gears.

As stated earlier, the author recommends dual path monitoring for acceleration sensors installed on critical gears. Filters used to separate the data paths should be easy to change in the field. Each data path should have two independently adjustable alarms.

If automatic shutdown is required on gears with casing sensors, two sensors and dual voting shutdown logic, or a several second shutdown delay, is recommended. This will prevent an automatic shutdown if someone accidentally strikes one sensor.

Temperature monitors employed on gears, or any critical machinery for that matter, should include all the applicable features of API 670 and 678. When Resistance Temperature Detector (RTD) sensors are used, the temperature monitor must be able to recognize an open sensor and disable the failed channel before it goes into alarm. Both RTD and thermocouple monitors should automatically drive downscale upon sensor failure.

What Might the Future Hold?

Everyone is continuing to look for an earlier warning of gear problems. Literature has been mentioned, and referenced in this paper, discussing the use of characteristics such as the harmonics of gear mesh frequency and sidebands, as measures of gear condition. Thus far there isn't any generally accepted method. A significant factor has been the lack of detailed historical information during the development of problems on in-service gears.

One hoped for advantage of multiple path monitoring is that changes in gear condition will be recognized earlier and documented. If there is a common factor, it can be identified and a more selective monitoring system designed. Thus, a lot is still unknown about gear monitoring, however, we can provide effective and reliable protection and give ourselves the means to recognize changes in condition far earlier. With this beginning, we may well find future systems monitoring some calculated or derived parameter with full confidence that it is the most accurate and responsive measure of gear condition.

LIMITS

To close this exposition on gear condition monitoring, we should briefly discuss vibration amplitude limits. AGMA Standard 426.01 contains guidelines for limiting vibration amplitudes on gears. [8] The vibration limits specified in AGMA Standard 426.01 are based on shaft displacement from 50HZ to 600 HZ. Limits, in mils, are defined by the expression

$$\sqrt{\frac{200}{f}}$$

where f is the frequency of vibration in HZ. Above 600 HZ the AGMA limit shifts to a constant casing acceleration of 10 g's peak.

To assess gear condition from casing vibration, the author uses a constant velocity criteria similar to that published and supplied with most portable instruments. In the author's experience, an overall casing velocity of .15 in/sec or less indicates good condition. From .15 to .35 in/sec is acceptable, becoming marginal. Above .35 in/sec is cause for concern. It should be emphasized that these are guidelines only. There are unquestionably numerous gears running "successfully" at velocity amplitudes in excess of .35 in/sec. Conversely, there probably have been cases where gears were in serious difficulty below .35 in/sec velocity.

At frequencies above 1,000 HZ the author's experience indicates gears must be treated individually. As mentioned earlier, unless amplitudes are obviously excessive, changes are far more important than absolute values.

CONCLUSION

The dynamics of gears are significantly different from centrifugal compressors and steam turbines. Because of the differences, the most effective methods for monitoring the condition of gears differ from those employed on compressors and steam turbines. Acceleration sensors mounted on the gear casing will pick up more condition-related information than other types of transducers. Properly utilized, the information obtained from one or more casing-mounted acceleration sensors will provide the most efficient and accurate machinery monitoring and protection.

ACKNOWLEDGEMENT

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